International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2012

Flow Boiling Heat Transfer and Pressure Drop of R1234ze(E) and R32 in a Horizontal Micro-Fin Tube

Daisuke Baba
baba@phase.cm.kyushu-u.ac.jp

Takafumi Nakagawa

Shigeru Koyama

Flow Boiling Heat Transfer and Pressure Drop of R1234ze(E) and R32 in Horizontal Micro-fin tube

Daisuke BABA[1*], Takafumi NAKAGAWA[1], Shigeru KOYAMA[2]

[1] Interdisciplinary Graduate School of Engineering Sciences, Kyushu University,
Kasuga-shi Fukuoka, Japan
Phone number: +81-92-583-7840, Fax number: +81-92-583-7833,
E-mail address: baba@phase.cm.kyushu-u.ac.jp

[2] Faculty of Engineering Sciences, Kyushu University
Kasuga-shi Fukuoka, Japan
Phone number: +81-92-583-7831, Fax number: +81-92-583-7833,
E-mail address: koyama@cm.kyushu-u.ac.jp

ABSTRACT

In the present study, the flow boiling heat transfer of R1234ze(E), R32 and zeotropic mixture R1234ze(E)/R32 (50/50 mass%) in a horizontal micro-fin tube is experimentally investigated to clarify the pressure drop and heat transfer characteristics. Tested micro-fin tube is made of copper, and its geometry is as follows, 6.00 mm in outer diameter, 5.21 mm in mean inside diameter, 0.26 mm in fin height and 20 degree in spiral angle. Experiments are conducted in the mass velocity range of 150 to 400 kg $m^{-2}s^{-1}$ at a constant inlet temperature 10 ^{o}C. The effects of mass velocity and the difference of refrigerants on heat transfer and frictional pressure drop characteristics are clarified. The measured pressure drop and heat transfer characteristics of pure refrigerants are also compared with some correlations.

1. INTRODUCTION

With the concern over the global warming and ozone layer depletion, developing environmentally friendly refrigerants for air conditioning and refrigeration systems is increasingly important. The key points of the development are to be low GWP (Global Warming Potential) and to perform COP equal to or higher than the currently used refrigerants such as R134a or R410A.

Ueda *et al*. (2011) experimentally compared the COP and capacity of R134a and R1234ze(E) with a 1407 kW class centrifugal chiller. The values of COP and capacity of R1234ze(E) were 3% and 29% lower than those of R134a, respectively. They concluded that this degradation of R1234ze(E) is still in adjustable range with compressor optimization. Koyama *et al*. (2011) evaluated a cycle performance of R410A, R1234ze(E), and zeotropic mixture R1234ze(E)/R32 with an experimental facility which performs 1.4 to 2.8 kW class heating and cooling mode air conditioning. With increasing mass fraction of R32, the COP and capacity were increased. COP and capacity of mixture almost reached that of R410A at R32 mass fraction 80 %. As evaluated, R32 is expected to increase the COP and capacity; however, the GWP is also increased. Based on their result, mixtures of R1234ze(E) and R32 are nominated as alternatives of R410A for residential air conditioners.

There are several studies on flow boiling heat transfer of low GWP refrigerants in horizontal tubes in the latest literatures. Regarding to R1234yf, Saitoh *et al*. (2011) were experimentally investigated flow boiling heat transfer coefficient of R1234yf in a horizontal smooth tube (I.D. 2 mm). The heat transfer coefficient (HTC) was measured at heat fluxes from 6 to 24 kWm^{-2}, mass velocities from 100 to 400 kg $m^{-2}s^{-1}$, and evaporating temperature 15 ^{o}C. The results show that the effect of heat flux and mass velocity on the heat transfer was large at high vapor quality. HTC of R1234yf was almost same as that of R134a. HTC calculated based on correlations with Saitoh *et al*. agreed well with the measured values, compared to other correlations. The measured pressure drop agreed well with that predicted by the Lockhart-Martinelli correlation. Li *et al*. (2011) were experimentally investigated flow boiling heat transfer coefficient of R1234yf/R32 mixtures in a smooth tube of 2 mm inner diameter. Concentrations of R1234yf are 80% and 50%. The test results show that HTC of the mixture with 20% mass fraction of R32 is lower than that of pure R1234yf. When the concentration of R32 increases to 50%, HTC of the mixture is higher than that of pure R1234yf at large mass velocities and high heat fluxes. HTC of pure R32, pure R1234yf and the mixtures predicted by six correlations were compared. It is indicated that the predicted results by the correlation for mixtures proposed in their study is in good agreement with measured HTC. Regarding to the other refrigerant R1234ze(E) and mixtures containing R1234ze(E), HTC data in smooth and micro-fin tubes seem not sufficient to clarify the effects of diameters and fin geometries on flow boiling heat transfer and pressure drop characteristics.

This study presents experimentally determined HTC and frictional pressure drop of pure refrigerants R1234ze(E), R32 and zeotropic refrigerant mixture R1234ze(E)/R32(50/50mass%) in a horizontal micro-fin tube of 6 mm O.D. In addition, the experimental data of HTC and frictional pressure drop for pure refrigerant are compared with some correlations proposed for other refrigerants flow boiling in micro-fin tubes.

2. EXPERIMENTAL APPARATUS, PROCEDURE AND CONDITIONS

Figure 1 shows an experimental apparatus used in this study. The apparatus is mainly composed of a variable speed compressor (1), an oil separator (2), a condenser (3), a liquid reservoir (4), a coriolis type mass flow meter (6), a solenoid expansion valve (7), a pre-heater (8), a test section (9) and an after-heater (10). Three temperature controlled bathes (11) supply hot water to the test section (9) and the after-heater (10), respectively. The rotational speed of the compressor and the opening angle of the expansion valve are manually controlled to adjust evaporating pressure and refrigerant mass flow rate. The pre-heater is controlled to adjust the test section inlet vapor quality.

Figure 2 shows dimensions of the test section and measurement points. The test section of 2216 mm total length is a tube-in-tube counter flow heat exchanger, in which the refrigerant flows in an inner test micro-fin tube and the heating water flows in the annulus constructed between inner and outer tubes. The annulus is divided into four subsections, each of which is 454 mm long (the active heating length is 414 mm). Two mixing chambers are installed at both ends of the test section to measure the refrigerant pressure and bulk mean temperature.

The maximum reading and measurement uncertainty of absolute and differential pressure transducers are 2MPa ±2.2 kPa, and 100 kPa ±200 Pa, respectively. The refrigerant temperatures at the test section inlet and outlet are measured with K-Type thermocouples with an uncertainty of ±0.05 K. The heating water temperatures at inlet and outlet of each subsection are measured with Pt thermometers within ±0.03 K uncertainty. T-type thermocouples of 0.127 mm in wire diameter are embedded in the outside of the inner test micro-fin tube on top, bottom, right, and left at axially middle points of each subsection. The measurement uncertainty of these thermocouples is ±0.05 K. The refrigerant mass flow rate is measured within ±0.12 % of full scale 13.8 g/s. The volumetric flow rate of water is measured by a gear type flow meter with the uncertainty ±0.5 % of the reading value.

Table 1 specifies the geometric parameters of the test micro-fin tube made of copper. Its mean inside diameter is 5.21 mm. The surface enlargement ratio of the test micro-fin tube is 2.55. This is the ratio of actual heat transfer area of micro-fin tube to that of 5.21 mm I.D. smooth tube. The present micro-fin tube, which is developed for high pressure refrigerants such as R410A used in residential air conditioners, has relatively high fins of 0.256 mm in height.

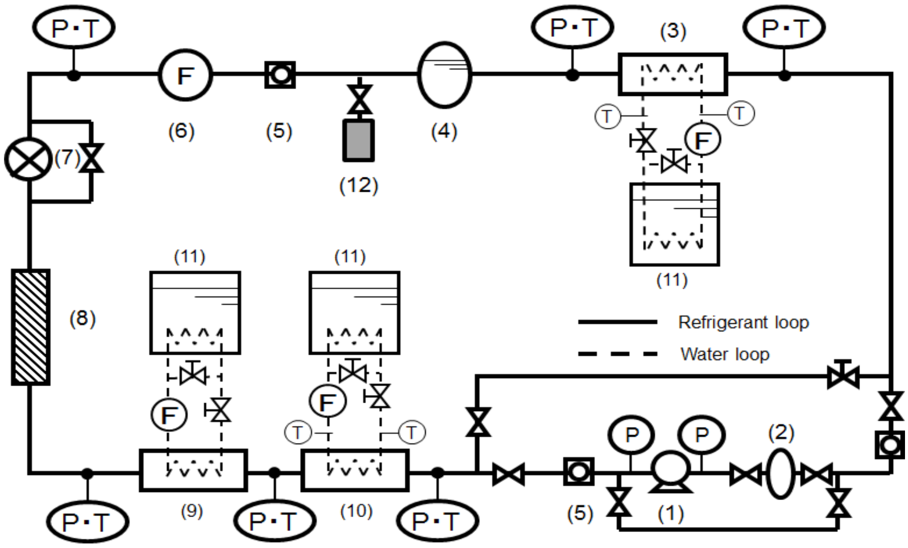
Experiments are conducted at the mass velocities ranging from 150 to 400 $\mathrm{kg\,m^{-2}s^{-1}}$, and the saturation temperature 10 ^{o}C. Experimental data on the steady state conditions are recorded every one second for one minute using a data acquisition system.

3. DATA REDUCTION

The frictional pressure drop ΔP_F is calculated as follows.

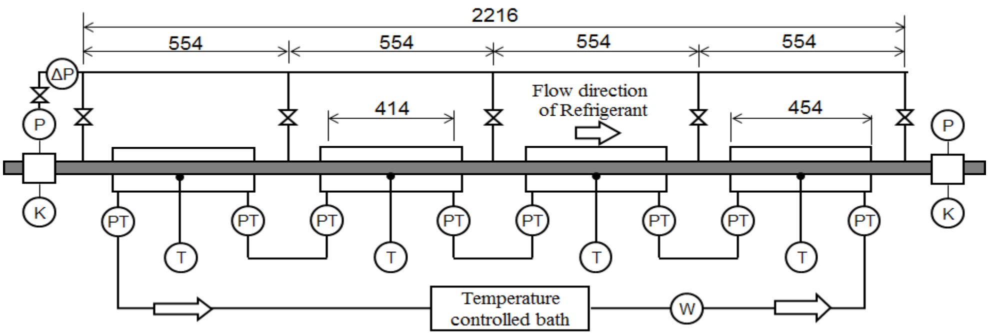
$$\Delta P_F = \Delta P_T - \Delta P_M \tag{1}$$

where ΔP_T is the measured total pressure drop, and ΔP_M is the acceleration pressure drop estimated from the following equation,



(1) Compressor
(2) Oil Separator
(3) Condenser
(4) Liquid Reservoir
(5) Sight Glass
(6) Flow Meter
(7) Expansion Valve
(8) Pre-Heater
(9) Test Section
(10) After-Heater
(11) Temperature Controlled Bath
(12) Sampling Port

Figure 1: Schematic view of the experimental apparatus



K : K-type Thermocouple
T : T-type Thermocouple
PT : Resistance Thermocouple
P : Absolute Pressure Transducer
Δ P : Differential Pressure Transducer
W : Gear Type Flow Meter

Figure 2: Schematic view of the test section

Table 1: Specification of the test micro-fin tube

Outside diameter d_o [mm]	**Max. inside diameter** d_r [mm]	**Mean inside diameter** d_i [mm]	**Height of fins** h_{fin} [mm]	**Spiral angle** β [deg.]	**Number of fins** N [-]	**Surface enlargement ratio** η [-]
6.05	5.37	5.21	0.256	18.8	58	2.55

$$\Delta P_{\mathrm{M}} = \Delta\left(\frac{G^2 x^2}{\xi \rho_{\mathrm{V}}} + \frac{G^2 (1-x)^2}{(1-\xi)\rho_{\mathrm{V}}}\right) \tag{2}$$

where x is the vapor quality, G is the mass velocity, and ξ is the void fraction that is obtained from the following correlation proposed by Koyama *et al.* (2001) for R134a in micro-fin tubes.

$$\xi = 0.81\xi_{\mathrm{Smith}} + 0.19x^{100(\rho_{\mathrm{V}}/\rho_{\mathrm{L}})}\xi_{\mathrm{Homo}} \tag{3}$$

$$\xi_{\mathrm{Smith}} = \left[1 + \left(\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{V}}}\right)\left(\frac{1-x}{x}\right)\left(0.4 + 0.6\sqrt{\frac{\dfrac{\rho_{\mathrm{V}}}{\rho_{\mathrm{L}}} + 0.4\dfrac{1-x}{x}}{1 + 0.4\dfrac{1-x}{x}}}\right)\right]^{-1} \tag{4}$$

$$\xi_{\mathrm{Homo}} = x\left[x + (1-x)\left(\frac{\rho_{\mathrm{V}}}{\rho_{\mathrm{L}}}\right)\right]^{-1} \tag{5}$$

where ξ_{Smith} is Smith's correlation (1971) and ξ_{Homo} is the void fraction of homogeneous two phase flow in smooth tubes, ρ_{V} and ρ_{L} are densities of saturated vapour and liquid, respectively.

The specific enthalpy at the after-heater outlet $h_{\mathrm{AH,out}}$ is determined from refrigerant pressure and bulk mean temperature measured at the mixing chamber. The specific enthalpy change through the after-heater Δh_{AH} is obtained from the water side heat balance. Hence, the specific enthalpy at the test section outlet $h_{\mathrm{TS,out}}$ is given as the subtraction of Δh_{AH} from $h_{\mathrm{AH,out}}$. The inlet bulk enthalpy in each subsection is calculated by solving the following heat balance equation from the test section outlet to upstream.

$$h_{\mathrm{b,in}} = h_{\mathrm{b,out}} - Q/W_{\mathrm{R}} \tag{6}$$

where $h_{\mathrm{b,in}}$, $h_{\mathrm{b,out}}$, Q, and W_{R} are subsection inlet and outlet bulk enthalpies, and the heat transfer rate calculated by water side and refrigerant mass flow rate, respectively. The refrigerant average vapor quality x given by

$$x = (h_{\mathrm{b}} - h_{\mathrm{L}})/(h_{\mathrm{V}} - h_{\mathrm{L}}) \tag{7}$$

where, h_{b} is the arithmetic mean of $h_{\mathrm{b,in}}$ and $h_{\mathrm{b,out}}$. h_{L} and h_{V} are enthalpies of liquid and vapor at the saturation point calculated from the arithmetic mean of pressures measured at inlet and outlet of each subsection and the composition of R1234ze(E)/R32 assumed uniform in vapour and liquid phases. The average heat flux based on the actual heat transfer area in each subsection is defined as follows.

$$q = Q/(\pi d_i \eta \Delta Z) \tag{8}$$

where d_{i}, η, and ΔZ are mean inside diameter, the surface enlargement ratio, and active heat transfer length, respectively. The circumferentially averaged internal tube wall temperature at the mean inside diameter T_{wi} is calculated as,

$$T_{\mathrm{Wi}} = T_{\mathrm{Wo}} + Q/(2\pi\lambda_{\mathrm{W}}\Delta Z)\ln(d_{\mathrm{o}}/d_{\mathrm{i}}) \tag{9}$$

where T_{wo}, λ_{w}, d_{o} are average temperature of the tube outer surface, thermal conductivity of the test tube and the outer diameter of the test tube, respectively. The average HTC based on actual heat transfer area in each subsection α is defined as,

$$\alpha = q/(T_{\mathrm{Wi}} - T_{\mathrm{b}}) \tag{10}$$

where T_b is the bulk temperature that is the equilibrium saturation temperature of refrigerant. For zeotropic mixture R1234ze(E)/R32, it is obtained from the enthalpy and pressure at the inlet and outlet of each subsection. For pure refrigerant R1234ze(E) and R32, T_b represents the saturation temperature. Properties of R1234ze(E), R32, R1234ze(E)/R32(50/50 mass%) and heating water were calculated with REFPROP Ver. 9.0 (2010) in this study. Properties of the zeotropic refrigerant mixture are calculated by circulating composition collected at sampling port.

4. RESULT AND DISCUSSION

4.1 Heat transfer characteristics

Figure 3, 4 and 5 show the HTC α of R1234ze(E), R32 and R1234ze(E)/R32(50/50 mass%) at saturation temperature 10 ^{o}C, and mass velocities G = 150, 200, 300 and 400 kg m^{-2}s^{-1}. The horizontal axis is vapor quality x. For pure refrigerant R1234ze(E) and R32, HTC increases with increasing vapor quality at any experimental conditions. This is mainly caused by thinning liquid film and faster vapor flow. For zeotropic refrigerant mixture R1234ze(E)/R32(50/50 mass%), HTC is almost unchanged with vapor quality change in the experimental range. That is thought to be calculated by the bulk enthalpy assuming the same temperature in liquid and vapor phase. For instance, HTC of low vapor quality is overestimated; while, HTC of high vapor quality is underestimated. Yoshida *et al*. (1993) stated that the HTC of zeotropic mixtures can be still enhanced at low vapor quality region, although the enhancement is greatly degraded. This degradation is theoretically explained by taking account of the local composition change in the meniscus liquid film formed in the fins on upper part of the tube perimeter. Similar to the Yoshida *et al*. (1993), HTC of mixture is lower than that of pure refrigerants at low vapor quality region.

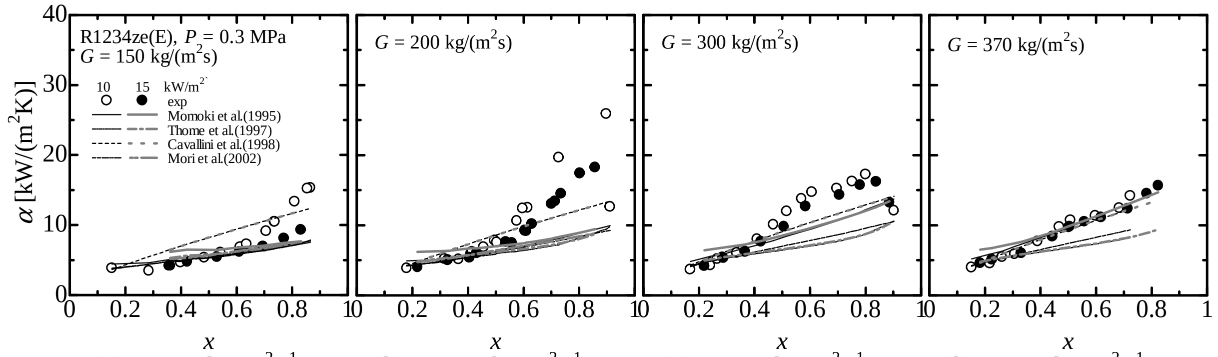
4.1.1 Comparison of HTC of various refrigerants: In comparison with between HTC of R1234ze(E) and R32, HTCs of R1234ze(E) are lower than those of R32 in the experimental mass velocity range. This can be explained with lower thermal conductivity and smaller latent heat of R1234ze(E). And also, HTC of zeotropic mixture R1234ze(E)/R32 is degraded from that of pure refrigerant at any condition because of the effect of mass transfer resistance. That is, individual components pass from the liquid to the vapor phase in different proportions, and the faster evaporation of the more volatile component causes an enrichment of the bubble forming boundary layer with the less volatile component, so that the local boiling temperature increases.

4.1.2 The effect of heat flux: For R1234ze(E) and R32, HTC decrease with increasing heat flux from 10 to 15 kWm^{-2} at vapor qualities beyond 0.7, and mass velocities below 200 kg m^{-2}s^{-1} as shown in Figures 3(a), 3(b), 4(a) and 4(b). That is the typical behavior of “dryout” occurrence at the top of tube. As shown in Figures 3(c), 3(d), 4(c) and 4(d), HTC is almost unchanged by heat flux. It appears to be that the convective evaporation heat transfer is dominant in those regions. For zeotropic refrigerant mixture, HTC increases a little with increasing heat flux from 10 to 15 kWm^{-2} in the entire range of vapor quality. The reason is that the recovery of nuclear boiling is supplied by abundant degreed of superheat for bubble formation with increasing heat flux against suppression of nuclear boiling due to the effect of mass transfer resistance.

4.1.3 The effect of mass velocity: As shown in Figures 3 and 4, the pure refrigerants show very slight increase in HTC with increasing mass velocity. Those HTCs increase with increasing vapor quality. The degree of this inclination suddenly becomes steep at a certain vapor quality. This suggests the transient of flow regime from stratified/wavy to annular by increasing vapor velocity, reducing liquid film thickness and enhancing convective heat transfer at vapor-liquid interface. This transient point seems to shift to lower vapor quality with increasing mass velocity. The annular flow seems to appear at lower vapor quality for higher mass velocity. HTC of the zeotropic refrigerant mixture also slightly increases with increasing mass velocity. However, the increasing ratio is still far below from that of pure refrigerants.

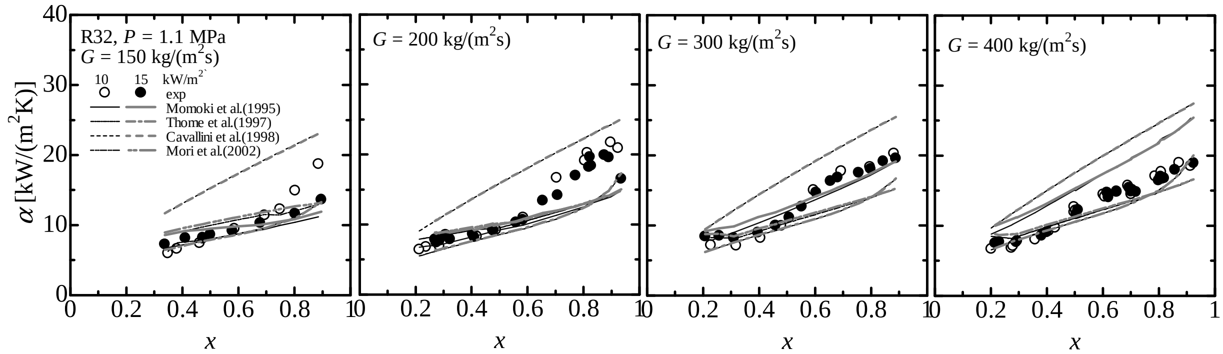
4.2 Frictional pressure characteristics

Figures 6, 7 and 8 show the frictional pressure drop ($\Delta P_F/\Delta Z$) of R1234ze(E), R32 and R1234ze(E)/R32(50/50mass%) at saturation temperature 10 ^{o}C and mass velocities 150, 200, 300 and 400 kg m^{-2}s^{-1}. Frictional pressure drops of all test refrigerants increase with increasing vapor quality and mass velocity. This can be explained with increasing shear stress at vapor-liquid interface and tube wall.



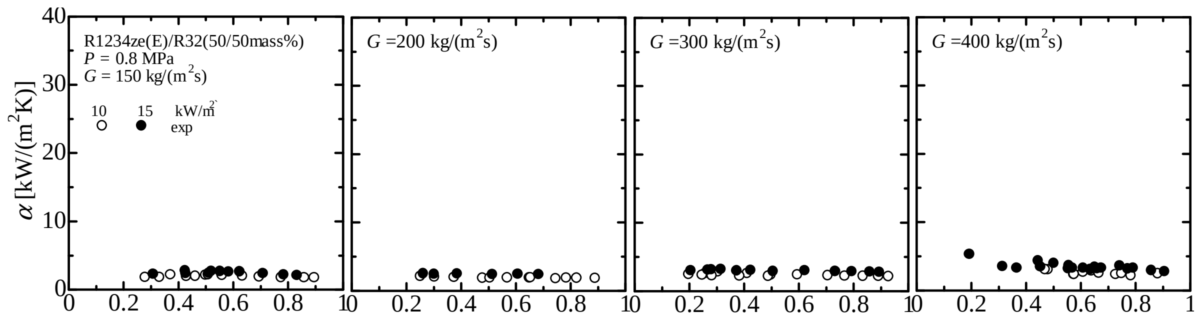
(a) G=150 kg m^{-2}s^{-1} (b) G=200 kg m^{-2}s^{-1} (c) G=300 kg m^{-2}s^{-1} (d) G=400 kg m^{-2}s^{-1}

Figure 3: HTC of R1234ze(E) with various mass velocities



(a) G=150 kg m^{-2}s^{-1} (b) G=200 kg m^{-2}s^{-1} (c) G=300 kg m^{-2}s^{-1} (d) G=400 kg m^{-2}s^{-1}

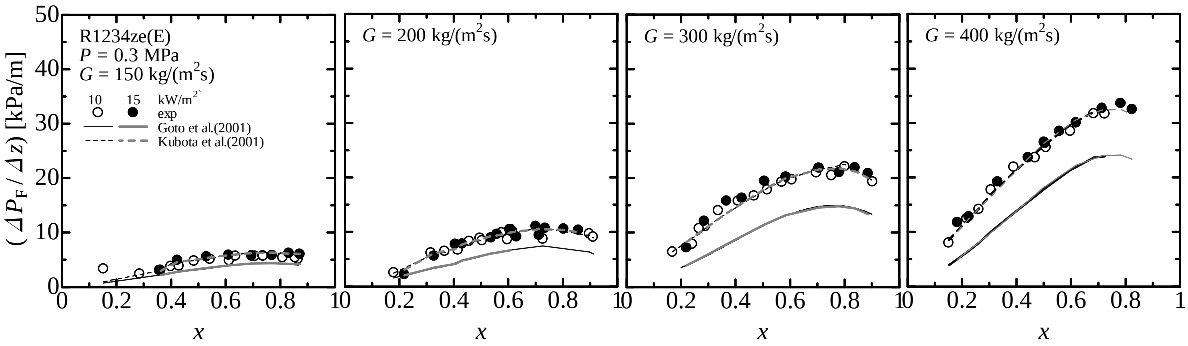
Figure 4: HTC of R32 with various mass velocities



(a) G=150 kg m^{-2}s^{-1} (b) G=200 kg m^{-2}s^{-1} (c) G=300 kg m^{-2}s^{-1} (d) G=400 kg m^{-2}s^{-1}

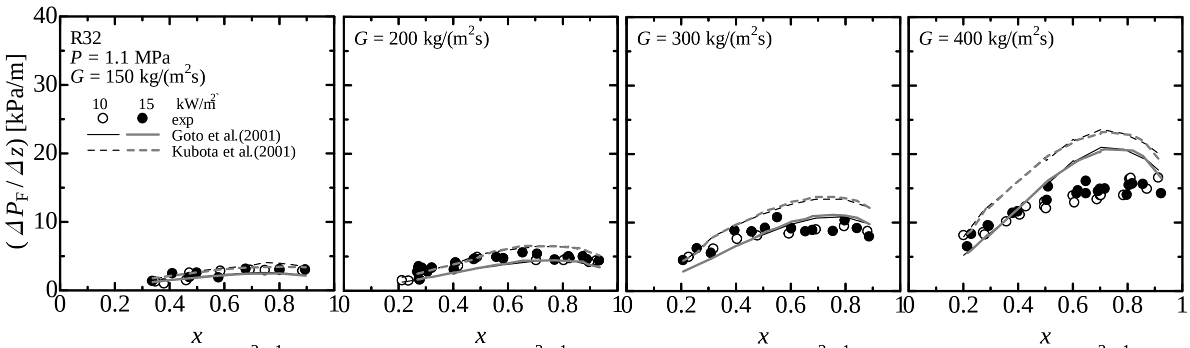
Figure 5: HTC of R1234ze(E)/R32(50/50mass%) with various mass velocities

4.2.1 Comparison of HTC of various refrigerants: As compared in Figures 6 and 7, frictional pressure drop of R1234ze(E) is higher than that of R32. At vapor quality 0.8 and mass velocity 300 kg m^{-2}s^{-1}, the frictional pressure drop of R1234ze(E) is twice of R32 roughly. This is because of more viscous liquid and higher vapor velocity due to the lower vapor density of R1234ze(E). Frictional pressure drop of zeotropic refrigerant mixture is higher than that of R32, lower than that of R1234ze(E). Because vapor density of mixture refrigerant is intermediate value between R1234ze(E) and R32.



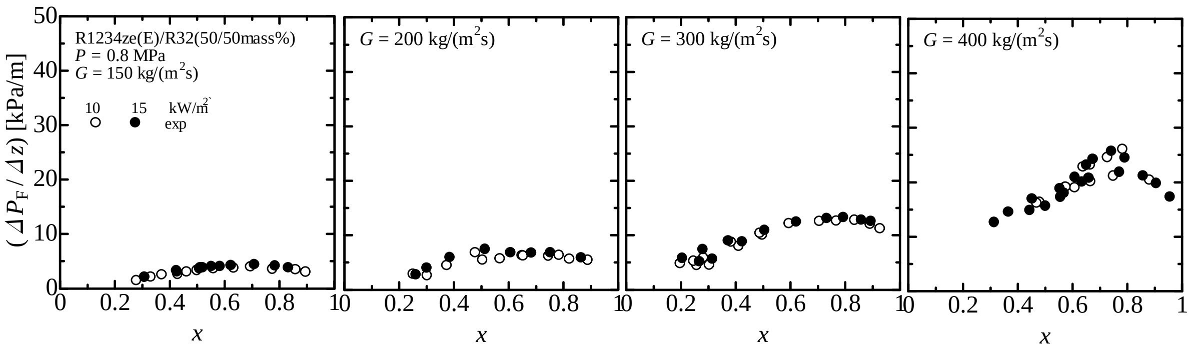
(a) G=150 kg m^{-2}s^{-1} (b) G=200 kg m^{-2}s^{-1} (c) G=300 kg m^{-2}s^{-1} (d) G=400 kg m^{-2}s^{-1}

Figure 6: Frictional pressure drop of R1234ze(E) with various mass velocities



(a) G=150 kg m^{-2}s^{-1} (b) G=200 kg m^{-2}s^{-1} (c) G=300 kg m^{-2}s^{-1} (d) G=400 kg m^{-2}s^{-1}

Figure 7: Frictional pressure drop of R32 with various mass velocities



(a) G=150 kg m^{-2}s^{-1} (b) G=200 kg m^{-2}s^{-1} (c) G=300 kg m^{-2}s^{-1} (d) G=400 kg m^{-2}s^{-1}

Figure 8: Frictional pressure drop of R1234ze(E)/R32(50/50mass%) with various mass velocities

4.3 Comparison between experimental data and correlations

4.3.1 Comparison between experimental data and predicting correlations on HTC: The present experimental results of pure refrigerants are compared with correlations proposed for other refrigerants by Momoki *et al*. (1995), Thome *et al*. (1997), Cavallini *et al*. (1998) and Mori *et al*. (2002). In Figure 3 and 4, black and grey lines show those predicting correlations. Table 2 compares the average deviation and mean deviation of those prediction results to experimental results. The predicted HTC of R1234ze(E) by above four correlations reasonably agree with the present experimental data with mean deviation 30%. For R32, the predicted HTC by the correlation of Thome *et al*. (1999) shows best agreement among the selected correlations with the present experimental data with mean deviation 14.8%. As compared lines and symbols in Figures 3 and 4, predicted HTC deviates from the present data mostly at low mass velocity and high vapor quality.

Table 2: Comparison of experimental HTC and selected correlations for R1234ze(E) and R32

Refrigerants	Momoki *et al.* (1995)		Thome *et al.* (1997)		Cavallini *et al.* (1998)		Mori *et al.* (2002)	
	AD[a]	MD[b]	AD[a]	MD[b]	AD[a]	MD[b]	AD[a]	MD[b]
R1234ze(E)	3.49	23.5	18.5	27.28	-10.4	23.9	15.7	24.1
R32	-9.9	20.5	12.5	14.8	-48.2	48.2	-13.8	31.9

a $\text{Average deviation} = \frac{1}{n}\sum_{1}^{n}(\alpha_{\text{EXP}} - \alpha_{\text{CAL}})/\alpha_{\text{EXP}} \times 100$

b $\text{Mean deviation} = \frac{1}{n}\sum_{1}^{n}\text{ABS}\ [(\alpha_{\text{EXP}} - \alpha_{\text{CAL}})/\alpha_{\text{EXP}}] \times 100$

Table 3: Comparison of frictional experimental pressure drop and selected correlations for R1234ze(E) and R32

Refrigerants	Goto *et al.* (2001)		Kubota *et al.* (2001)	
	AD[a]	MD[b]	AD[a]	MD[b]
R1234ze(E)	34.2	34.2	0.51	11.2
R32	-0.26	21.5	-12.7	16.9

a $\text{Average deviation} = \frac{1}{n}\sum_{1}^{n}(\Delta P_{\text{EXP}} - \Delta P_{\text{CAL}})/\Delta P_{\text{EXP}} \times 100$

b $\text{Mean deviation} = \frac{1}{n}\sum_{1}^{n}\text{ABS}\left[(\Delta P_{\text{EXP}} - \Delta P_{\text{CAL}})/\Delta P_{\text{EXP}}\right] \times 100$

4.3.2 Comparison of frictional pressure drop between experimental data and correlations: The present experimental results on frictional pressure drop for pure refrigerants in the micro fin tube are compared to the correlations of Goto *et al*. (2001) and Kubota *et al*. (2001) as shown with black and grey lines in Figures 6, 7. Table 3 compares average and mean deviation of those two correlations from the present data. For both pure refrigerants, the predicted pressure drop by the correlation of Kubota *et al*.(2001) shows better agreement with the present experimental data. The prediction results agreed with present data with mean deviation of 11.2% for R1234ze(E), and 16.9% for R32.

5. CONCUSIONS

The characteristics of the frictional pressure drop and the HTC of boiling flow are experimentally investigated for R1234ze(E), R32 and R1234ze(E)/R32(50/50mass%) in a horizontal micro-fin tube at the mass velocities ranging from 150 to 400 kg $m^{-2}s^{-1}$, and constant inlet temperature 10 ^{o}C. The main conclusions are summarized as:

(1) HTC of R1234ze(E) is lower than that of R32. This can be explained with lower liquid thermal conductivity and smaller latent heat of R1234ze(E). HTC of the zeotropic mixture is degraded from that of pure refrigerant, because of mass transfer resistance.
(2) The frictional pressure drop of R1234ze(E) is higher than that of R32. This is because of more viscous liquid and higher vapour velocity of R1234ze(E). Frictional pressure drop of R1234ze(E)/R32(50/50mass%) is higher than that of R32, but lower than that of R1234ze(E). Because the refrigerant properties are intermediate value between R1234ze(E) and R32.
(3) The predicted HTC by Momoki *et al*. (1995), Cavallini *et al*. (1998), Thome *et al*. (1999) and Mori *et al*. (2002) correlation of R1234ze(E) agree with the present experimental data within 30%. For R32, the values predicted by the correlation of Thome *et al*. (1999) are in good agreement with the present experimental data with mean deviation 14.8%.
(4) The predicted frictional pressure drop by the correlation of Kubota *et al*.(2001) agrees well with the present experimental data within 11.2% for R1234ze(E), and 16.9% for R32.

NOMENCLATURE

d_i	mean inside diameter	(m)
d_o	outside diameter	(m)
d_r	maximum inside diameter	(m)
G	mass velocity	(kg m^{-2}s^{-1})
h	specific enthalpy	(J kg^{-1})
h_{fin}	height of fins	(m)
N	number of fins	(–)
q	average heat flux	(W m^{-2})
Q	heat transfer rate	(W)
T	temperature	(^{o}C)
W_{R}	refrigerant mass flow rate	(kg s^{-1})
x	vapor quality	(–)
ΔP_{F}	frictional pressure drop	(Pa)
ΔP_{M}	acceleration pressure drop	(Pa)
ΔP_{T}	total pressure drop	(Pa)
Δz	tube length between pressure ports	(m)
ΔZ	active heat transfer length	(m)
α	heat transfer coefficient	(W m^{-2} s^{-1})
β	spiral angle	(degree)
η	surface enlargement ratio	(–)
λ	thermal conductivity	(W m^{-1} K^{-1})
ρ	density	(kg m^{-3})
ξ	void fraction	(–)

Subscripts

b	bulk
L	liquid
V	vapor
in	in
out	out
W	wall
Wi	inner wall
Wo	outer wall
Homo	homogeneous
Smith	Smith's model
EXP	experiment
CAL	calculation

REFERENCES

Cavallini, A., Del Col, D., Longo, G A., Rossetto, L, 1998, Refrigerant vaporization inside enhanced tubes: a heat transfer model, *Proceeding of Heat Transfer in Condensation and Evaporation: Application to industrial and environmental processes: Eurotherm 62 Seminor*: p. 222-231.

Goto, M., Inoue, N., Ishikawa, N., 2001, Condensation and evaporation heat transfer of R410A inside internally grooved horizontal tube,, *Int. J. Refrig*., vol. 24: p. 628-638.

Koyama, S., Chen, S., Kitano, R., Kuwahara, K., 2001, Experimental study on void fraction of two-phase flow inside a micro-fin tube, *The Reports of Institute for advanced Material Study, Kyushu University*, vol. 15, no. 1: p. 79-85.

Koyama, S., Takata, N., Fukuda, S., 2011, An experimental study on heat pump cycle using zeotropic binary refrigerant of HFO-1234ze(E) and HFC-32, *10th IEA Heat pump Conference 2011*., ID 6.5

Kubota, A., Uchida, M., Shikazono, N., 2001, Predicting Equations for Evaporation Pressure Drop Inside Horizontal Smooth and Grooved Tubes , *Trans. JSRAE*., vol. 18, no. 4: p. 393-401 (in Japanese).

Lemmon, E.W., Huber, M.L., McLinden, M.O., 2010, *Reference Fluid Thermodynamic and Transport Properties-REFPROP*, Version 9.0, National Institute of Standards and Technology, Gaithersburg.

Li, M., Dang, C., Hihara, E., 2011, Study of flow boiling heat transfer characteristics of low GWP refrigerants in a smooth tube, *Pcoc. 48th HRSJ*.

Momoki, S., Yu, J., Koyama, S., Fujii, T., Honda, H., 1995, A Correlation for Forced Convective Boiling Heat transfer of Refrigerants in a Microfin Tube, *Trans. JSRAE*., vol. 12, no. 2: p. 177-184 (in Japanese).

Mori, H., Yoshida, S., Koyama, S., Miyara, A., Momoki, S., 2002, Prediction of heat transfer coefficients for refrigerants flowing in horizontal, spirally grooved evaporator tubes, *Proceeding of 2002 JSRAE*: p. 547-550 (in Japanese).

Saitoh, S., Dang, C., Nakamura, Y., Hihara, E., 2011, Boiling heat transfer of HFO-1234yf flowing in a smooth small-diameter horizontal tube, *Int. J. Refrig*., vol. 34, no. 8: p. 1846-1853

Smith, S.L., 1971, Void fractions in two phase flow: A correlation based on an equal velocity heat model, *Int. J. Heat Fluid Flow*, vol. 1, no. 1: p. 22-39.

Thome, J., Kattan, N., Favrat, A., 1997, Evaporation in microfin tube: A generalized prediction model, *Proceeding of Convective Flow and Pool Boiling Conference*: p. 239-244.

Ueda, K., Wajima, K., Yokoyama, A., Shimizu, A., 2011, Study of application Low GWP refrigerant to centrifugal chiller Evaluation of performance for centrifugal chiller with HFO-1234ze(E), *Proceeding of 2011JSRAE*: p. 199-202 (in Japanese).

Yoshida, S., Hong, H., Mori, H., 1993, Enhancement of Heat Transfer to a Non-Azeotropic Refrigerant Mixture in a Horizontal, Spirally Grooved Evaporator Tube, *Trans. JSRAE.*, vol. 59: p. 283-288 (in Japanese).

ACKNOWLEDGEMENT

The present study was sponsored by the project on “The Development of Non-fluorinated Energy-saving Refrigeration and Air Conditioning System” of New Energy and Industrial Technology Development Organization, Japan. The authors express gratitude here to Dr. Chieko Kondo who gave useful advice about the present study.